Fermi National Accelerator Laboratory

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Component Exergy Analysis of a Liquid Helium Refrigerator Upgrade

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COMPONENT EXERGY ANALYSIS OF A LIQUID HELIUM REFRIGERATOR UPGRADE

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ABSTRACT

Plans call for upgrading the existing Fermilab liquid helium satellite refrigeration system by adding cold compressors with an accompanying phase separator return dewar. Accelerator performance will be enhanced by further lowering superconducting magnet temperatures. Two possible configurations for utilizing the stored refrigeration in the liquid dewar were studied: 1) precooling the wet expander inlet and 2) aftercooling the wet expander exhaust. A second law exergy analysis, which quantifies each source of irreversibility as a power input to the cycle, was performed to provide a comparison in operating costs between these two schemes. Aftercooling the expander exhaust results in a 1% gain in second law efficiency over the precooling configuration, due primarily to the more efficient use of the dewar liquid.

NOMENCLATURE

DE = destroyed exergy
E = exergy
e = exergy per unit mass
E_{in} = exergy carried into control volume
E_{out} = exergy carried out control volume
E_Q = exergy carried into control volume by heat transfer
E_W = exergy carried into control volume by work
H = enthalpy
h = enthalpy per unit mass
$\dot{m}$ = flow rate
S = entropy
s = entropy per unit mass
T = temperature
W_{rev} = ideal reversible work

Note: a dot above a quantity indicates "rate of" that quantity

Introduction

The Fermilab satellite refrigeration system [1] and the Central Helium Liquefier (CHL) [2] provide cooling for approximately 1000 superconducting magnets and assorted cryogenic components that make up the Tevatron particle accelerator. The current system is capable of maintaining peak magnet temperatures at about 4.9K, allowing the Tevatron to operate reliably at a beam energy of 900 GeV. One aspect of Fermilab's upgrade plans involves the installation of cold compressors to further lower magnet temperatures and allow for higher beam energy [3].

The cryogenic system upgrade requires that the cold compressors with an accompanying phase separator return dewar be fitted into existing refrigerator building piping. The phase separator dewar protects the cold compressor from possible damage due to liquid surges and provides a source of stored refrigeration available from the liquid in the dewar. Two flow configurations were considered to utilize this stored refrigeration: 1) precooling the wet expander inlet and 2) aftercooling the wet expander exhaust. Figures 1 and 2 illustrate these configurations. Each schematic represents the layout of one of the 24 refrigerator buildings in the complete satellite refrigeration system.

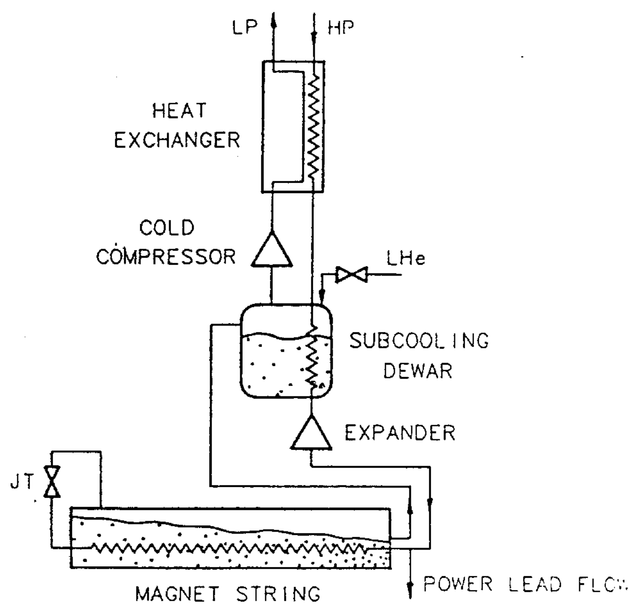

Figure 1: Precooling Configuration

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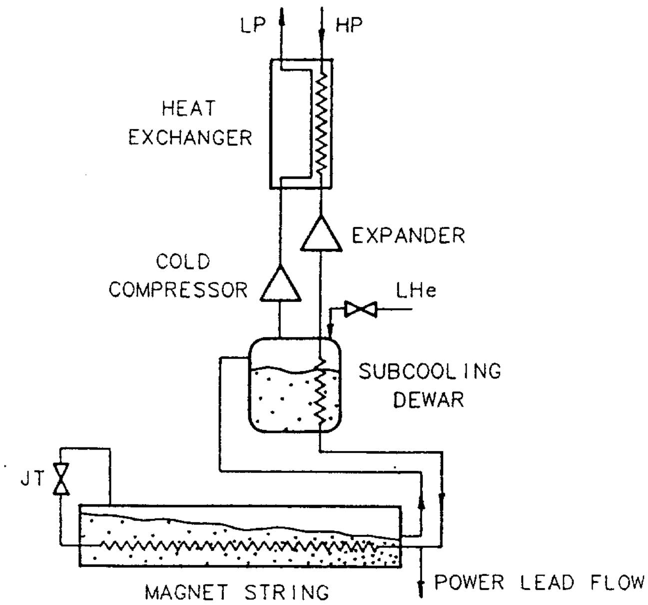

Figure 2: Aftercooling Configuration

In Figures 1 and 2, the satellite refrigerators are boosted by taking additional refrigeration from the CHL to provide the heat exchanger with a sufficient flow imbalance to cool the incoming high pressure flow. This normal operation mode is referred to as "satellite mode." An alternative way of providing the heat exchanger with the necessary imbalance is to operate in "mixed mode." This mode reduces the amount of liquid required from the CHL by gaining refrigeration from expanding at an intermediate point in the heat exchanger column and precooling some of the high pressure helium inlet flow in a liquid nitrogen exchanger. The "mixed mode' heat exchanger configuration is illustrated in Figure 3. Both precooling and aftercooling configurations can be operated in either satellite mode or mixed mode.

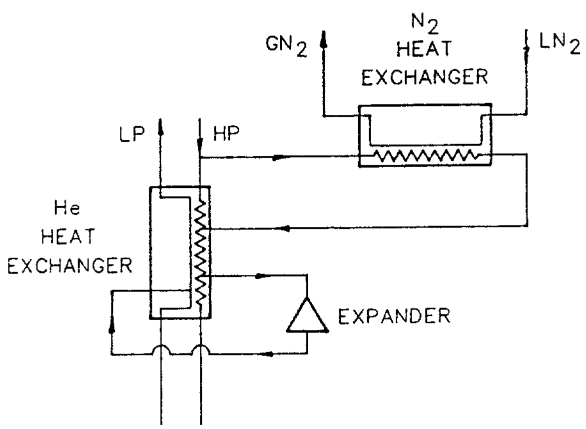

Figure 3: "Mixed Mode" Heat Exchanger Operation

To provide a thermodynamic assessment of these configurations and operating modes, a second law analysis for the steady state case was performed. The second law is useful in assessing thermodynamic cycles by allowing sources of irreversibility to be quantified. An exergy analysis is a particularly useful form of second law analysis since it equates these irreversibilities to power input to the cycle, a direct measure of operating cost. The exergy analysis will be done on a component by component basis. Exergy losses will quantify the irreversibilities in each refrigerator component in a particular configuration.

Cycle Calculation Description

Satellite refrigeration performance was simulated using the assumptions and operating conditions of Table 1. Helium properties were evaluated from work by Arp and McCarty [4]. Matter and energy balances were used on each component to completely define state points and flow rates. The helium heat exchanger column was integrated into the cycle with a finite difference solution in the extremely cold end of the exchanger (to account for the large helium property variations in this region) and an effectiveness-NTU solution elsewhere. With all state points and flow rates known, component power requirements and heat transfer rates can be predicted.

Table 1: Cycle Assumptions
(For typical refrigerator building in the Tevatron satellite refrigeration system)

Specified operation conditions:
Magnet heat load = 700W
Dewar two phase temp. = 3.5K
(corresponding two phase press. = 0.46 atm)
Magnets return 2 phase mixture with 20% liquid to dewar
Precooler outlet temp. difference = 0.5K
Aftercooler outlet temp. difference = 0.1K

General process assumptions:
Wet expander isentropic efficiency = 75%
Cold compr isentropic efficiency = 60%
CHL Carnot efficiency = 17%
Warm compr isothermal efficiency = 40%
All single phase piping press. = 2 atm
All two phase piping press. = dewar press.
Total lead flow = 1 g/s
Heat exchr hi press tube inlet = 20 atm, 300K
Heat exchr low press shell inlet = 1.2 atm
No pressure drop across exchanger.
CHL provides liquid He at 3 atm, 5.05K.
Single phase temp. constant through magnets.
Magnet temp. 0.2K greater than 2 phase temp.

Additional process assumptions for mixed mode operation:
Dry expander isentropic efficiency = 75%
Helium mass flow through nitrogen exchanger = 4.2 g/s
Liquid nitrogen into exchanger at 3 atm, 80K.
Gaseous nitrogen leaves exchanger at 230K
Helium leaves nitrogen exchanger at 82K

Exergy Analysis

A popular modern method of analyzing a thermodynamic cycle is to define a quantity of "exergy" at each state point [5,6,7]. Exergy is defined as the maximum amount of useful energy of a system in its environment when bringing the system from any given state to a state of complete equilibrium with the environment. For a state of constant composition and steady flow:

$$H_1 - H_0 - T_0(S_1 - S_0) = \text{"exergy of state 1"} = E_1 \qquad (1)$$

where 0 denotes ambient conditions of the environment and positive work is into the system. This equation can be expressed on a rate basis as well:

$$\dot{m}\left[h_1 - h_0 - t_0(s_1 - s_0)\right] = \dot{m}e_1 \qquad (2)$$

By using exergy to express a state point in terms of work (or power) available or required, a convenient measure of value is obtained.

This concept can be extended to particular processes by considering the overall exergy change as found from "exergy balancing" [7]. By accounting for all exergy flowing into and out of some control volume, the "destroyed exergy" can be calculated. This destroyed exergy represents irreversibilities in the process which are now expressed in units of value significance, such as Watts. Then, sources of irreversibilities can be pinpointed and the cost of these irreversibilities can be easily judged. For a steady state process in a control volume, an exergy balance gives:

$$DE = E_{in} - E_{out} + E_W + E_Q \tag{3}$$

where DE is destroyed exergy, E_{in} is the exergy carried in by the process fluid, E_{out} is the exergy carried out be the process fluid, E_W is exergy carried in by work, and E_Q is exergy carried in by heat transfer. For an ideal process, DE equals zero; therefore, DE is always greater than or equal to zero.

Consider what is represented by E_W and E_Q and their signs. The useful energy in a quantity of work is simply the work itself. Therefore, the exergy of a quantity of work is :

$$E_W = W \tag{4}$$

Since positive E_W is defined as exergy carried in, this term is positive when work is added. The exergy of a quantity of heat transferred at a constant temperature is given by the Carnot work either required to produce or available from the heat transfer process:

$$E_Q = Q(T - T_0)/T \tag{5}$$

where positive Q is heat added to the control volume and T is the temperature from which Q is being transferred. When the heat transfer requires refrigeration, E_Q is negative; when the heat transfer can run a heat engine, E_Q is positive. Substituting for E_W and E_Q in (eqn 3):

$$DE = E_{in} - E_{out} + W + Q(T - T_0)/T \tag{6}$$

On a rate basis:

$$D\dot{E} = \dot{m}_{in}\, e_{in} - \dot{m}_{out} e_{out} + \dot{W} + \dot{Q}(T - T_0)/T \tag{7}$$

As an example, consider a case of interest in cryogenics: extracting heat of refrigeration Q from a temperature region T with $T < T_0$. Assume no work is involved. Referencing (eqn 6) shows that the E_Q term is negative and E_{in} will be greater than E_{out}. This indicates that some exergy is accounted for by the work required to operate a Carnot refrigerator extracting Q at temperature T. If the process is not doing refrigeration equal to a Carnot cycle, losses will result.

Results

Since the satellite refrigerator simulation has determined all process state points, exergy values for each point can be calculated. Then, the exergy balancing concepts can be used to compute the rate of exergy destruction in each refrigerator component. The Central Helium Liquefier is considered to be another component of the satellite refrigerator; its losses are based on the plant operating at an assumed second law efficiency. The rate of exergy addition to the refrigeration system is equal to the power required to run the warm and cold compressors and CHL, less the expander power produced.

Magnet refrigeration and power lead cooling results from the useful exergy remaining. For the magnet strings, this "flow" of useful exergy amounts to the Carnot power required to refrigerate the magnet heat load at the specified magnet operating temperatures. The useful exergy flow to the power leads is taken to be the exergy rate convected in by the lead flow.

Having defined for the system the total exergy input rate, the exergy destruction rate of each component, and the useful flow of exergy produced, one can generate a graphical depiction of exergy flow. When using "pie" charts, the complete pie represents the total exergy (or, equivalently, power) input to the cycle. This exergy flowing into the system is then divided into portions representing either useful exergy flows or exergy losses. The portions are identified in terms of both power and percent of total power input. Note that the percentage of useful flow represents a second law efficiency. An ideal Carnot cycle would have all of its input exergy converted to useful exergy with no losses.

Figures 4 and 5 show the exergy flow charts for the precooling wet expander case and the aftercooling wet expander case operating in satellite mode. The aftercooling wet engine configuration is seen to be superior in efficiency. It would operate at an efficiency of about 14% while the precooler configuration would give about 13%. From a power usage standpoint, the aftercooling setup provides the same amount of cooling but requires about 8% less power. (This 8% reduction in required power occurs because the aftercooling configuration is able to satisfy the cycle refrigeration requirements with roughly this percentage reduction in flow rates). The exergy flow diagrams show that a smaller percentage of exergy is destroyed in the dewar when its used for aftercooling.

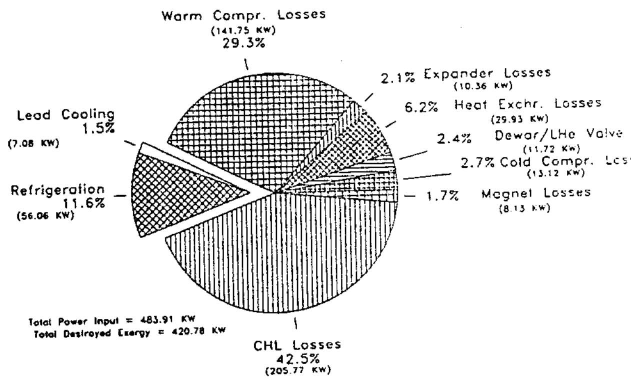

Figure 4: Exergy Flow for Precooling Configuration

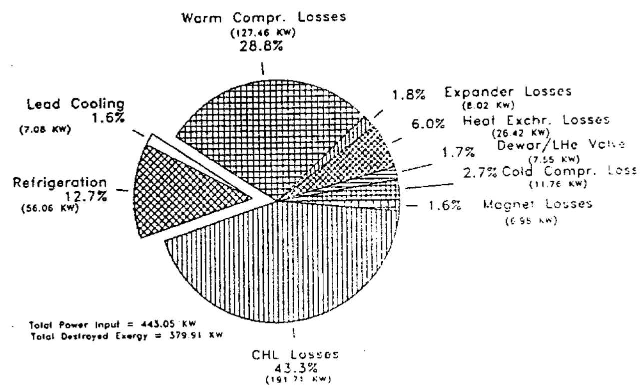

Figure 5: Exergy Flow for Aftercooling Configuration

The aftercooling configuration offers the more thermodynamically efficient use of the liquid thermal buffer since it adds heat to the dewar at the lower temperature (the expander outlet temperature as opposed to the expander inlet temperature). This fact is recognizable before performing a second law analysis; however, the second law analysis allows the gains in thermodynamic efficiency to be quantified. These quantified efficiency gains from aftercooling can be more clearly judged against advantages offered by the precooling arrangement (such as slower expander speeds due to increasing inlet fluid density). Another point that the exergy flow charts reveal is that the aftercooling configuration gains some in efficiency by having a larger percentage of losses accounted for in the more efficient CHL. Finally, this analysis identifies the warm compressors as the satellite system component that generates the most inefficiency. Thus, any attempts to improve the satellite system efficiency would profit most from warm compressor efficiency improvements.

Since aftercooling was identified as the more efficient configuration, it was further analyzed to judge its performance when operating in mixed mode with the heat exchanger utilizing an intermediate expander and a nitrogen precooling exchanger. The exergy flow chart in Figure 6 reveals that this operating mode is slightly less efficient than running the aftercooling configuration in satellite mode. CHL now accounts for a smaller percentage of overall losses since less helium flow is required from this source. The warm compressor accounts for a larger percentage of losses since the diversion of some of the refrigerator flow through the heat exchanger column expander leads to greater compressor flow rates. Efficiency performance is hurt by accounting for more losses in the relatively inefficient warm compressors as opposed to the relatively more efficient CHL plant. Also, the heat exchanger loss percentage increases. The exchanger is now forced to handle more mass flow and thus accounts for losses that were previously assigned to the CHL. Furthermore, although there is an efficiency benefit from reducing the temperature difference along the length of the heat exchanger with the expander, the quite large temperature differences in the nitrogen exchanger add to the exchanger losses.

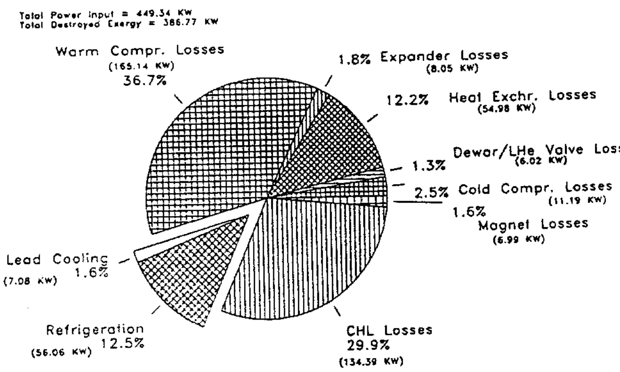

Figure 6: Exergy Flow for Aftercooling Configuration with Exchanger Operating in Mixed Mode.

Conclusions

A second law–exergy analysis gives a powerful way to compare two possible refrigeration cycle configurations. Areas of irreversibilities can be identified, and a cost can be attributed to these component losses. Aftercooling the wet expander exhaust results in a 1% gain in efficiency over the precooling configuration, due primarily to the more efficient use of the dewar liquid. No gain in efficiency results in operating the heat exchanger in the mode that uses an intermediate expander and nitrogen precooling.

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